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Design of a suspended handle to attenuate rock drill hand-arm vibration: model development and validation

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Abstract

Pneumatic jackleg drills are widespread percussion tools used in the mining industry. Hand-arm vibration frequency-weighted exposure levels evaluated within the 6.3–1250 Hz frequency range have been found to be on the order of 25 m/s^2 when operating this tool in typical mining conditions. This study concerns the development of a suspended handle designed to provide attenuation of the vibration and shocks being generated at the tool blow frequency, which occurs between 35 and 45 Hz on most types of pneumatic drills. The results of the first phase of development of such a handle are presented in this paper. They are based on the development and validation of a model combining the suspended handle and the hand-arm system. For that purpose, the hand-arm system is represented by a four-degree-of-freedom lumped parameter model, referred to as model 2 in the ISO 10068 standard. As part of this investigation, a model is developed to represent two different types of suspended handles: one incorporating helicoidal springs, the other viscoelastic mounts. These combined hand-arm-suspended handle models are then validated by comparing the model predictions with the measurements of vibration transmissibility realized while exciting the suspended handles on an electrodynamic shaker system. These measurements involved the use of human subjects holding the handles while applying a push force varying from 0 to 80 N, and a grip force ranging from 20 to 50 N. For values of grip and push forces set at 50 N, good agreement was achieved between the model predictions and the measurements, especially at frequencies above 35 Hz. The hand was found to have a significant influence on the vibration transmissibility responses of both suspended handles. When the values of suspension stiffness were selected to provide a resonant frequency of 25 Hz for the free handles, the vibration attenuation achieved at a frequency of 35 Hz was on the order of 30% when gripping the handle, while a slight amplification was noted at that frequency with the free handle. As the frequency increased towards 45 Hz, even more important attenuation could be realized (on the order of 50%) for the handle-hand combination. When the resonant frequency of the free suspended handle was set

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at a higher frequency (i.e., 67 Hz), the hand-arm system was found to provide additional damping at the handle resonant frequency, while not introducing any significant influence at the lower frequencies as compared with the behavior observed for lower natural frequency suspended handles.

1. Introduction

Pneumatic jackleg drills are widespread percussion tools used in the mining industry. They are used when the dimensions of the galleries do not allow large size automatic extraction machines to operate inside the mines. Jackleg drills differ from most other light portable power tools by the fact that the feed or push force is applied through the retractable leg. This does not prevent the operator, however, from having to maintain his hands in contact with the handle of the machine, where the controls, including that of the leg are positioned. The grip and push forces are estimated to be less than 50 N during the drilling phase while they could reach values as high as 100 N for grip and 200 N for push when the miner starts a new hole. Hand-arm vibration frequency weighted r.m.s. accelerations of as much as 25 m/s^2 [1–4] have been reported on the handle of such tools over the 6.3–1250 Hz frequency range.

Prolonged exposure to hand-arm vibration is known to be associated with the development of peripheral vascular (i.e., Raynaud's phenomenon), neurological and musculo-skeletal disorders collectively referred to as the ''hand-arm vibration syndrome'' (HAVS).

The vibration produced by jackleg drills are predominant along the percussion axis, and the levels have been reported [1–3] to be at least 3 times lower along the transverse directions. The weighted r.m.s. vibration frequency spectrum as measured on the handle of such tools is found to have a dominant component arising at the drill's blow frequency, which usually occurs between 35 and 45 Hz on most types of pneumatic drills. Since the rotation of the drill bit is realized by an indexed rotational device driven by the reciprocating hammer, the operational rotational frequency is similar to the percussion frequency. Several technical solutions have been proposed over the years to reduce the vibration exposure levels on jackleg drills, but these have generally not been successful. Anti-vibration gloves have been proposed but these have not been shown to be effective for attenuating vibration below 150 Hz, corresponding to the range within which the weighted dominant frequencies appear during jackleg drill operations [5]. Suspended handles have also been developed based on passive elements, but these have generally been shown not to be efficient for attenuating vibration energy below 100 Hz or not resistant enough to withstand the severe conditions encountered during underground mining operations [6–9]. As for dynamic damping devices and active vibration control, such means have been considered but are still at present at the conceptual stage for applications on jackleg drills.

The above considerations have motivated the need to design a new anti-vibration system based on a robust spring–mass system tuned to attenuate vibration at the blow frequency of the tool. Considering that the weight of a pneumatic jackleg drill is already quite important, it was deemed necessary to impose a constraint to ensure that the design of the new suspended handle would not increase the mass of current tools. Furthermore, the suspension was required to be effective under push forces of as much as 200 N, as encountered when starting a new hole. The target on the amount of vibration attenuation that such a handle should provide was set at 50% in an attempt to maximize the likelihood of reducing the occurrence of HAVS amongst the population of miners. Such constraints involved selecting the best design characteristics to provide a compromise between reduced suspended handle mass while offering maximum hand-arm vibration attenuation. For that purpose, it was considered relevant to investigate the dynamic behavior of the combined hand-arm-suspended handle system, as a means of identifying the conditions and design parameters most likely to lead to increased attenuation performance while maintaining the handle mass to a minimum.

The aim of this paper is to present the results of the first phase of the study by focussing on the dynamic behavior of the combined system comprising the suspended handle coupled with the hand-arm. For that purpose, a model of the hand-arm system as given in the ISO 10068 standard is combined with models representing two types of suspended handles. These models are then validated experimentally on the basis of base-to-handle vibration transmissibility characteristics measured while subjects are gripping instrumented suspended handles mounted on an electrodynamic shaker system. The influence of loading the handles with the hand while varying the grip and push forces is then evaluated experimentally to examine their effect on the vibration transmissibility characteristics of the suspended handles. These results and the model predictions are then used to identify the most favorable suspended handle design characteristics needed to maximize the vibration attenuation on jackleg drills.

2. Model development

2.1. General description

A model is proposed to simulate the hand-arm system coupled with a suspended handle. Since the vibration is dominant along the percussion axis, the model is limited to this single axis. It is considered to apply only to account for the vibration response at frequencies below 500 Hz. The design of the suspended handle is based on the requirement to provide vibration attenuation at the drill blow frequency of 35 Hz. At such a low frequency, the handle is considered to act as a rigid mass, while the suspension is represented by massless springs and dampers. Furthermore, the model is assumed to behave linearly.

In the case of simple single-degree-of-freedom mechanical systems, the analysis of the motion associated with the basic components (mass, spring, damper) can be performed through the use of a four-pole representation [10]. Such poles establish the relationship between the forces and velocities at the input of a system with those at the output. This four-pole representation is retained for the analysis of the hand-arm model, which corresponds to the four-degree-of-freedom hand-arm model referred to as model 2 in the ISO 10068 standard [11,12]. The four-pole representations applicable to the handle (rigid mass), the handle suspension (spring–damper combination) and the hand-arm system (four mass, spring and damper combination) are developed below as a function of the angular frequency ω, where $\omega = 2\pi f$, and f represents the frequency.

2.2. Handle representation

For a rigid mass M, the force and velocity F_e, V_e at the input of the four-pole system are related to the force and velocity F_s, V_s at the output by the following relations: $V_e = V_s$ and $F_e =$

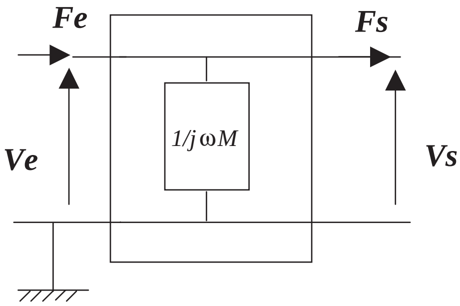

Fig. 1. Four-pole system representation of the handle.

$F_s + M_{\gamma_s}$, where $\gamma_s = \mathrm{j}\omega V_s$. The four-pole mass system representing the handle is shown in Fig. 1. The equations of motion applicable to this four-pole system are represented as

$$\begin{Bmatrix} F_e \\ V_e \end{Bmatrix} = \begin{bmatrix} 1 & \mathrm{j}\omega M \\ 0 & 1 \end{bmatrix} \begin{Bmatrix} F_s \\ V_s \end{Bmatrix}. \tag{1}$$

2.3. Handle suspension representation

Two types of suspension design are considered for the handle. The first type represents the handle suspension as a parallel combination of a massless spring with a viscous damper for which the damping coefficient is C. In this case, the equations relating the input force and velocity F_e and V_e of the four-pole system to those at the output, F_s and V_s, are given by

$$F_e = F_s = K(x_s - x_e) + C(V_s - V_e), \quad V_e = V_s + F_e/(C + K/\mathrm{j}\omega), \tag{2}$$

where K represents the spring stiffness coefficient and x_e and x_s represent the input and output displacements, respectively.

The four-pole representation of the suspension realized by combining a massless spring and a viscous damper is shown in Fig. 2 and the equations of motion are given as

$$\begin{Bmatrix} F_e \\ V_e \end{Bmatrix} = \begin{bmatrix} 1 & 0 \\ \mathrm{j}\omega/(K + \mathrm{j}\omega C) & 1 \end{bmatrix} \begin{Bmatrix} F_s \\ V_s \end{Bmatrix}. \tag{3}$$

The second type of suspension considers that the damping is structural and that it can be represented by the loss factor η. For this particular case, the input/output force and velocity relationships are given by the expressions

$$F_e = F_s = (x_s - x_e)K(1 + \mathrm{j}\eta), \quad V_e = V_s + \mathrm{j}\omega F_e/(K(1 + \mathrm{j}\eta)). \tag{4}$$

The four-pole representation for this type of suspension involving structural damping is shown in Fig. 3 for which the equations of motion are given as

$$\begin{Bmatrix} F_e \\ V_e \end{Bmatrix} = \begin{bmatrix} 1 & 0 \\ \mathrm{j}\omega(K(1 + \mathrm{j}\eta)) & 1 \end{bmatrix} \begin{Bmatrix} F_s \\ V_s \end{Bmatrix}. \tag{5}$$

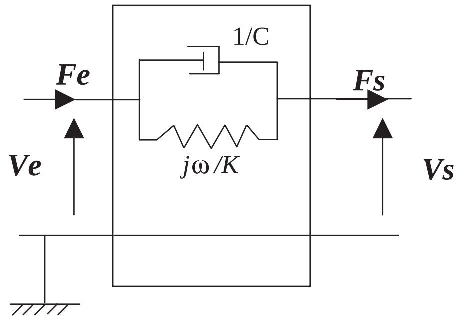

Fig. 2. Four-pole system representation of a massless spring and a viscous damper.

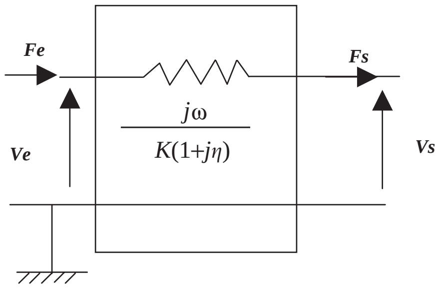

Fig. 3. Four-pole system representation of a massless spring and structural damping.

2.4. Hand-arm system representation

A four-degree-of-freedom linear model referred to as model 2 in ISO 10068 standard is selected to represent the hand-arm system. This model was derived to match measured driving-point mechanical impedance characteristics considered to apply under a specific set of conditions when the hand is gripping a cylindrical handle [11,12]. Furthermore, this model is considered to apply within the 10–500 Hz frequency range, and for hand grip forces within the range from 25 to 50 N and push forces less than or equal to 50 N. For the model to apply, the handle diameter must lie between 19 and 45 mm and the postural conditions applying to the wrist and the hand must conform to those described in the ISO 10068 standard [12].

The model parameters of the four-degree-of-freedom hand-arm system are selected as those which apply to the vibration acting along the z_h direction, corresponding to the dominant vibration axis when operating a jackleg drill. This direction corresponds to the axis of the forearm and is given by the axis z_h represented in the basicentric co-ordinate system defined in the ISO 5349-1 standard [13]. The hand-arm model parameters as given in the ISO 10068 standard for z_h axis vibration are reproduced in Table 1. The total mass of the hand is considered to be 5.06 kg on the basis of this model. The equivalent representation of the four-pole system, combining the suspended handle and the hand-arm system, is given in Fig. 4.

Table 1
Four-degrees-of-freedom hand-arm model parameters as defined in the ISO 10068 standard

$M_1 = 1.9 \times 10^{-2}$ kg	$K_1 = 3 \times 10^5$ N/m	$C_1 = 5.91 \times 10^2$ N s/m
$M_2 = 9.47 \times 10^{-2}$ kg	$K_2 = 6.8 \times 10^4$ N/m	$C_2 = 2.03 \times 10^2$ N s/m
$M_3 = 6.55 \times 10^{-1}$ kg	$K_3 = 1.99 \times 10^2$ N/m	$C_3 = 1.99 \times 10^2$ N s/m
$M_4 = 4.29$ kg	$K_4 = 2.04 \times 10^3$ N/m	$C_4 = 2.39 \times 10^2$ N s/m

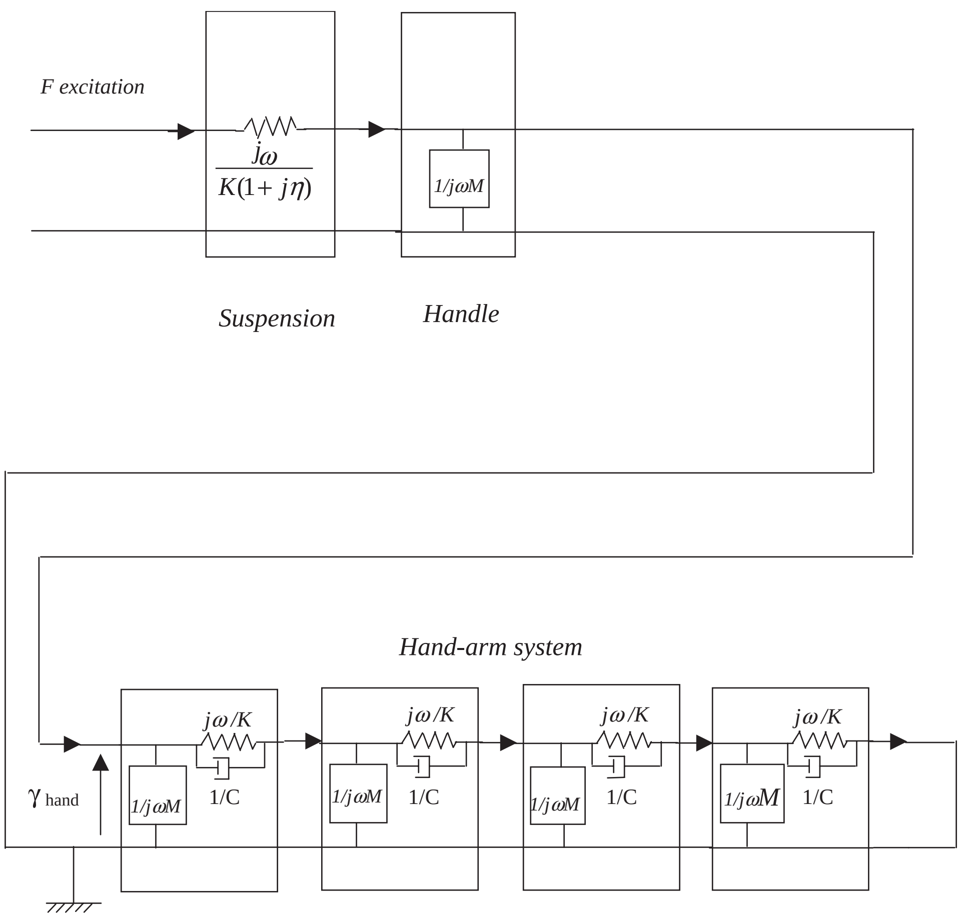

Fig. 4. Four-pole system representation of the suspended handle and the hand-arm system.

3. Suspended handle configuration

A model was developed to compute the optimal stiffness characteristics that would be needed to attenuate the drill handle vibration at the percussion frequency (35 Hz) through a suspension inserted between the hand and the handle. On the basis of this base-excited system, the computed value of the stiffness needed to provide 50% of vibration attenuation while taking into account the hand-arm system was 4.6×10^4 N/m.

Two different handle suspension configurations were considered: one realized with a parallel combination of four helicoidal springs, the other consisting of four viscoelastic mounts. As the stiffness of most commercially available springs and viscoelastic mounts is quite varied, only those components were selected which could provide stiffness values close to the desired values. Considering the tolerance given by the suppliers on the reported values, the true values of stiffness were measured for each spring or mount. For the springs selected, the static stiffness values given by the manufacturer were 1.26×10^4 N/m, providing an equivalent stiffness coefficient of 5.04×10^4 N/m for the parallel combination of four springs. These values were verified experimentally by measuring the static force-deflection characteristics of each spring under compression for two different deflections (2 and 4 mm) while using a static force sensor (Sensortronics 60001A1K-1000). The measured values for the four springs varied between 1.06×10^4 and 1.4×10^4 N/m, providing a mean estimated equivalent total static stiffness of 4.8×10^4 N/m for the four springs parallel combination.

The second type of suspension was realized by incorporating four viscoelastic mounts in parallel between the base and the handle, thus operating in shear mode. The static shear stiffness of the mounts reported by the manufacturer was 1×10^4 N/m, representing a total equivalent shear stiffness of 4×10^4 N/m for the four parallel mounts. The dynamic shear stiffness was not reported by the manufacturer and could not be measured with the instrumentation in our laboratory. However, some measures of dynamic compression stiffness were performed for the parallel combination of four mounts over the 30–300 Hz frequency range. The dynamic stiffness was obtained by measuring the ratio of the force transmitted to the ground by the mount to the displacement when excited from above by an electrodynamic shaker. The results showed that the dynamic compression stiffness was relatively constant within this frequency range and that its value was approximately 20% higher than the static stiffness. On that basis, it was estimated that the dynamic shear stiffness of the four mounts would follow a similar trend as the static shear stiffness, thus yielding an estimated value of 4.8×10^4 N/m for the dynamic shear stiffness. The damping ratio was estimated to be on the order of 0.09 over the 30–300 Hz frequency range when operating in shear mode.

4. Experimental setup

4.1. Description of the set-up

The experimental set-up is shown in Fig. 5. It consists of a horizontally mounted electrodynamic shaker (Unholtz Dickie TA250-S032) holding an aluminum handle of 45 mm diameter. The handle was isolated from the base plate by inserting either the springs or viscoelastic mounts in between. The suspended handle was fixed to the shaker head through a force sensor (Transducer Technique MLP-200-CO) to allow the measurement of the static push force. The grip force was measured with a strain gauge inserted between the two parts of a split handle.

A miniature single axis accelerometer (B&K 4374) inserted within the handle provided the measure of vibration at the hand–handle interface while another accelerometer (B&K 4393) measured the base plate acceleration. This base plate was excited by a broadband random white noise excitation with the power spectral density (PSD) magnitude fixed at 0.4 $(\mathrm{m/s^2})^2$/Hz between

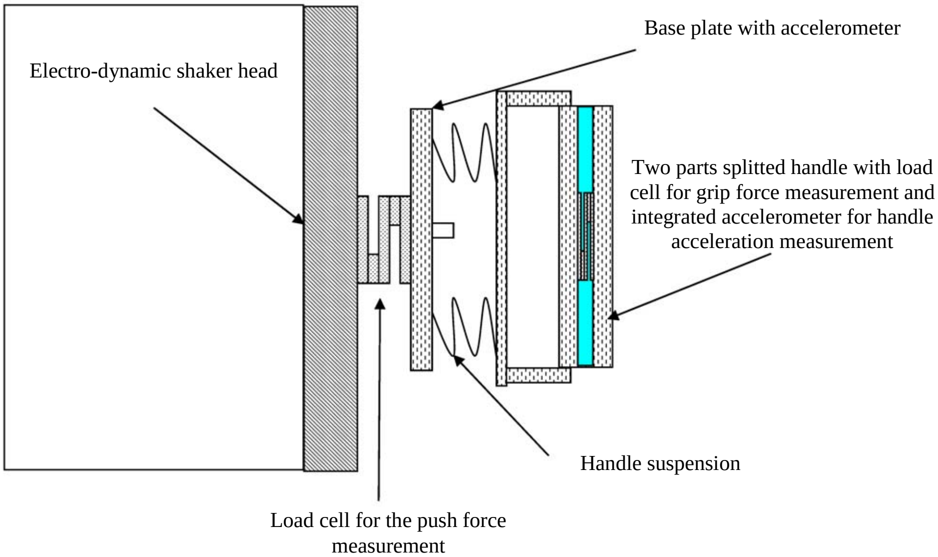

Fig. 5. Laboratory experimental set-up.

10 and 500 Hz, providing a r.m.s. acceleration of $14\,m/s^2$. For model validation purposes, only two right-handed male subjects took part in the experiments (subject 1: height 1.80 m and mass 80 kg; subject 2: height 1.75 m and mass 75 kg). Both subjects were instructed to stand in front of the experimental set-up and grasp the handle, while adopting a posture as defined in the ISO 10068 standard [12]. The position of the upper body was perpendicular to the axis of the shaker, while the forearm was oriented along the axis of the shaker. For both operators, the hand almost totally covered the cylindrical handle while ensuring that no overlap of the fingers would occur.

5. Hand-arm apparent mass characterization

5.1. Hand-arm model validation

Mathematically, the apparent mass is defined as the ratio of the dynamic force $F(\mathrm{j}\omega)$, applied at the point of input of vibration to the resulting acceleration $a(\mathrm{j}\omega)$, expressed by

$$M(\mathrm{j}\omega) = \frac{F(\mathrm{j}\omega)}{a(\mathrm{j}\omega)}. \tag{6}$$

The measurement of hand-arm apparent mass response function was performed in two steps as described below.

5.1.1. Handle apparent mass measurements

This step involved the determination of the apparent mass response function of the instrumented handle for the purpose of cancelling its contribution from that of the combined hand-arm–handle system when performing the measurements with the subjects while gripping the handle. The apparent mass of the handle was determined by measuring the force to acceleration transfer function of the handle when subjected to vibration without the hand. The modulus of the apparent mass response function was found to be constant and equal to 1.5 kg in the 10–400 Hz frequency range, above which it was observed to decrease with increasing frequency. The measured value was confirmed by weighing the handle, the mass of which was established to be 1.53 kg.

Fig. 6 presents the real and imaginary parts and the modulus of the apparent mass response of the free handle over the 10–400 Hz frequency range. These results reveal that the real part of the function coincides with that of the modulus over the entire frequency range considered.

5.1.2. Evaluation of the apparent mass of the hand-arm system

This step involved the determination of the apparent mass response of the hand combined with the handle and the computation of the apparent mass of the hand alone. The measurements were performed by requesting the subjects to grip the vibrating handle while exerting controlled values of grip and push forces and measuring the force to acceleration transfer function at the hand–handle interface. The apparent mass of the hand-arm system alone was computed by subtracting real and imaginary parts of the handle response function from those of the combined hand–handle combination.

The measurements of hand-arm apparent mass were realized with two test subjects while maintaining the push force constant at 50 N. In view of the necessity to use a rigid handle to measure hand-arm apparent mass, the strain gauge design of the current handle could not be used

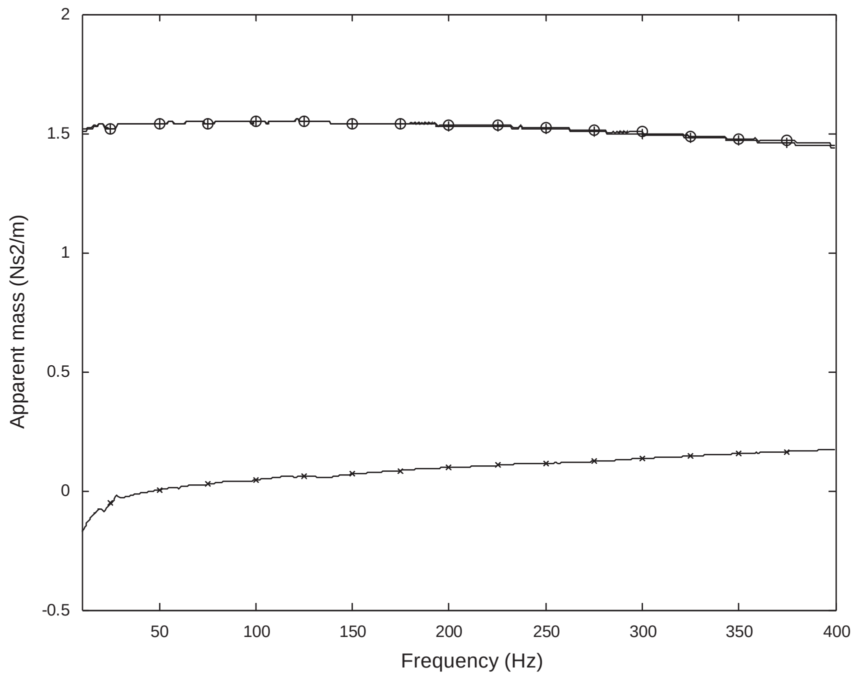

Fig. 6. Rigid handle apparent mass measured without the hand (+, real part; ×, imaginary part; ⊖, modulus).

to measure grip force directly during the measurements. Consequently, the grip force had to be estimated empirically by requesting each subject to train to apply the required grip force. The value of applied grip force was established to be 40 N for subject 1, and 60 N for subject 2, providing a mean estimated grip force of 50 N for the two subjects.

Fig. 7 presents the real and imaginary parts and the modulus response of the corrected hand-arm apparent mass function measured for subjects 1 and 2, respectively, while exerting a push force fixed at 50 N and a target grip force set at 50 N.

Below 50 Hz, the real part provides most of the contribution to the apparent mass modulus, showing a peak at 16 Hz for subject 1 and at 22 Hz for subject 2. Some differences are observed between measured and computed responses below 50 Hz which could perhaps be attributed to the difficulties for the two subjects selected to represent closely the mean biodynamic response of the larger population of subjects that the model defined in the ISO 10068 standard is intented to represent.

When averaging the apparent mass responses of the two subjects, the trends in behavior between the measured and computed responses are found to be very similar, as illustrated in Fig. 8. The mean measured apparent mass response generally falls within the range of idealized values defined in the ISO 10068 standard over most of the frequency range considered. The measurements show a peak occurring at low frequency around 20 Hz which is not present in

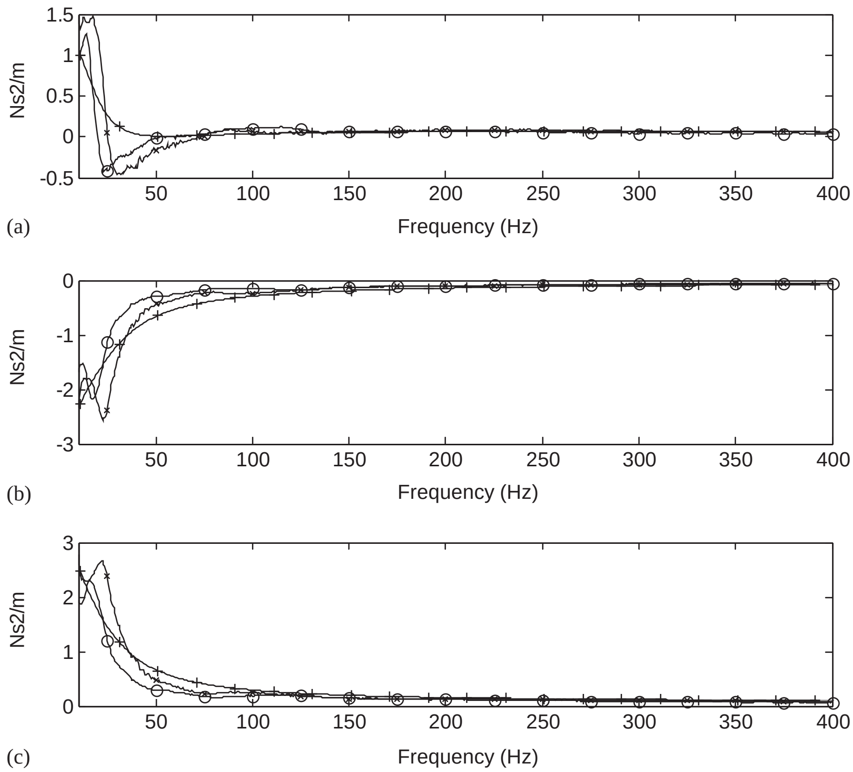

Fig. 7. Apparent mass of the hand-arm system: theory versus measure. Push force set at 50 N (+, computed; ×, subject 1; o, subject 2). (a) Real part, (b) imaginary part, (c) modulus.

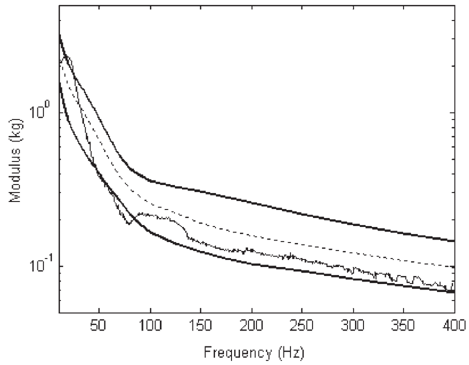

Fig. 8. Modulus of mean measured (subjects 1 and 2) and ISO 10068 apparent mass of the hand-arm system grip and push forces set at 50 N (——, ISO upper and lower limit; - - -, ISO median value; —, mean measured).

the mean model response given by the data in the ISO 10068 standard. As the four-degree-of-freedom model presented by the ISO 10068 standard is based on data obtained for a large number of subjects, the possibility exists that the low-frequency peak is perhaps diluted in the model simulation due to the use of averaged data on this larger subject population.

The results obtained in this part of the study suggest that the model accounts reasonably well for the hand-arm vibration apparent mass response measured under the conditions investigated in this study while the grip and push forces are maintained at 50 N.

These results further show that the apparent mass of the hand-arm system is most significant at frequencies below 50 Hz, where it reaches a peak of 2 kg at 20 Hz. At frequencies above 50 Hz, the apparent mass decreases drastically where it reaches a value of 500 g at 50 Hz, while falling below 200 g at frequencies above 80 Hz.

5.2. Influence of grip force on apparent mass response

Three levels of grip force referred to as low, medium and high were applied during these tests. These levels were subsequently measured and determined to correspond to 5, 40 and approximately 100 N for subject 1, and 5, 60 and 100 N for subject 2, respectively. In view of the difficulties associated with maintaining the grip forces equal to these values during the whole series of measurements, the evaluation of the influence of grip force on apparent mass response was performed only to establish overall trends rather than precise measurements at specific values of grip force.

An increase in grip force results in a shift towards higher frequencies of the minimum of the real part and of the peak of the modulus of apparent mass. This is illustrated in Fig. 9 which presents the hand-arm apparent mass frequency response over the 10–400 Hz frequency range as measured for subject 1 for different values of grip force while maintaining the push force constant at 50 N. Although not presented, the results showed the same trends for subject 2 and agree with the tendencies reported in Ref. [14]. The effect of increasing the grip force could be considered to be that of increasing the stiffness of the hand and forearm, thus causing a shift of the frequency of peak modulus response towards higher frequencies. At the same time, the apparent mass of the hand would however be expected to decrease with increasing frequency but this effect would not

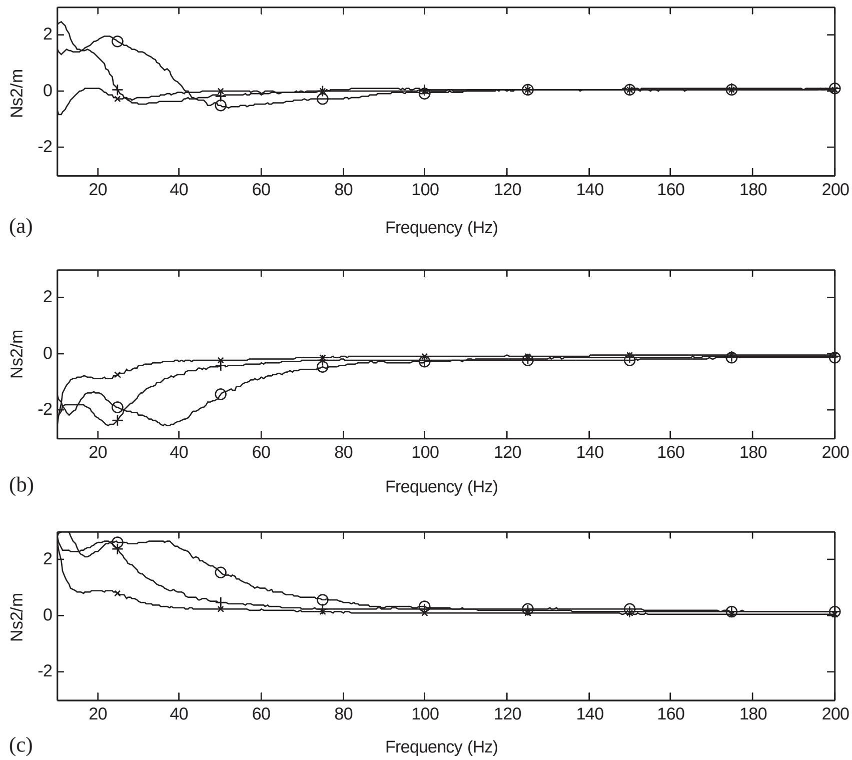

Fig. 9. Influence of the grip force on the apparent mass of the hand-arm system. Push force set at 50 (×, low grip; +, medium grip; o, high grip). (a) Real part, (b) imaginary part, (c) modulus.

be as significant as that provided by the increase in grip force, thus resulting in an upward shift of the peak modulus response. Not only is there an upward shift in the frequency of the peak modulus response with increases in grip force, but the peak modulus response is also observed to increase. Although the results are presented for grip forces up to 100 N, it was noted that the subjects had difficulties to maintain a steady grip over extended periods of time when it exceeded a value of 50 N. Furthermore, the model proposed in ISO 10068 and retained in this study is only considered valid for values of grip and push forces below 50 N.

6. Vibration transmissibility of suspended handles

6.1. Measurement method

The vibration transmissibility frequency response functions of the handles were determined by measuring the ratio of acceleration frequency spectra on the suspended handles to that at the base plate. As reported in Section 4, the acceleration vibration excitation at the base plate was provided by an electrodynamic shaker system generating a white noise vibration frequency spectrum with PSD fixed at 0.4 $(m/s^2)^2/Hz$ between 10 and 500 Hz.

Handle acceleration transmissibility measurements were performed for the free handles and when the hand was gripping the handles while applying fixed values of push and grip forces. This was realized for the two types of suspension involving helicoidal springs and viscoelastic mounts and with the two subjects whose characteristics were defined previously.

6.2. Hand–handle model validation

The combined suspended handle–hand-arm model validation was realized for both handles by comparing the computed transmissibility response function with mean measured responses obtained with the two subjects gripping the handles while applying push and grip forces set at 50 N.

Fig. 10 presents the results of such validation for the case involving the use of a parallel combination of four helicoidal springs inserted between the handle and the base. The transmissibility response function computed for the free handle is also shown in this figure. The stiffness and damping model parameters of the handle suspension were tuned so as to allow the computed model response to match the acceleration transmissibility response measured for the free handle. Under these conditions, the damping ratio and the equivalent stiffness of the four parallel spring combination were established as 0.031 and 44×10^3 N/m, respectively. These values are close to those which were measured directly, namely 48×10^3 N/m for the stiffness while damping was considered negligible.

From the response curves in Fig. 10 applying to the hand–handle combination, the shift in resonant frequency and the increase in damping introduced by loading the suspended handle with the hand appears to be well accounted for by the model. At frequencies above 25 Hz, both mean measured and computed transmissibility responses are in agreement. The attenuation is established as 30% at 35 Hz and 50% at 45 Hz on the basis of both measured and computed responses. The model appears to underestimate the magnitude of measured

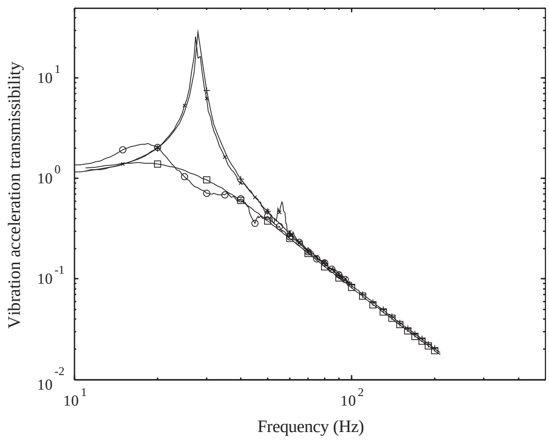

Fig. 10. Comparison of the mean measured and calculated vibration transmissibility response of a suspended handle (four helicoidal springs, +, calculated without hand; ×, measured without hand; □, calculated with hand; o, measured with hand).

transmissibility within the resonant frequency range. The most important deviation between computed and measured responses observed at frequencies below 20 Hz could possibly be reduced if the comparison could be based on mean measured data involving a larger population of subjects.

Fig. 11 presents the validation results for the case where the suspension consists of viscoelastic mounts. On the basis of the measured transmissibility response for the free handle, the parameters of the suspended handle model for this suspension were adjusted to provide a damping ratio of 0.09 and a suspension stiffness of $K = 46 \times 10^3$ N/m, thus close to the measured values reported in Section 3. The results are very similar to those which were reported for the parallel helicoidal spring combination, except that with the viscoelastic mounts, the model underestimates the suspended handle transmissibility response at frequencies below 25 Hz, while providing an overestimation within the 25–45 Hz frequency range. Above 45 Hz, both computed and mean measured responses agree very well. Overall, the extent of the deviations between model and measurements appears to be slightly more important with the viscoelastic mounts than with the helicoidal springs due to the non-linear behavior of the mounts.

Fig. 12 presents the results applicable to the case where the handle suspension is designed with four parallel helicoidal springs providing a global stiffness of 2.6×10^5 N/m, thus giving a free natural frequency of the suspended handle of 67 Hz. With this handle, both measured and computed transmissibility magnitudes are still in agreement. When the hand is placed in contact with the handle, however, the shift of the resonant peak towards lower frequencies which was apparent with lower natural frequency handles is found to have disappeared with this new design. Consequently, this handle cannot provide any vibration attenuation at the frequency of 35 Hz. Loading of the handle with the hand is thus seen to introduce additional damping at the higher natural frequency of this modified handle while not having any significant influence at lower frequencies due to the apparent mass of the hand arm system which becomes negligible above 50 Hz.

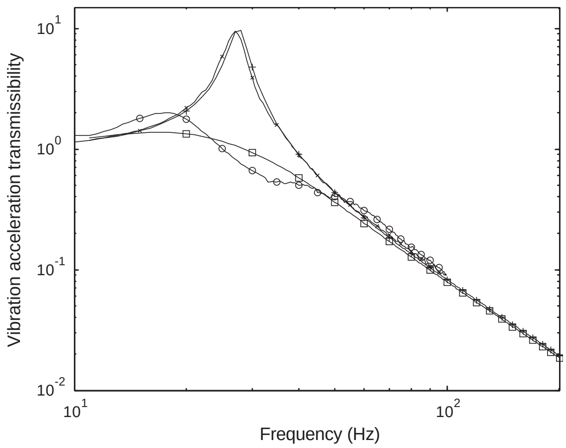

Fig. 11. Comparison of the mean measured and calculated vibration transmissibility response of a suspended handle (four viscoelastic mounts; +, calculated without hand; ×, measured without hand; □, calculated with hand; o, measured with hand).

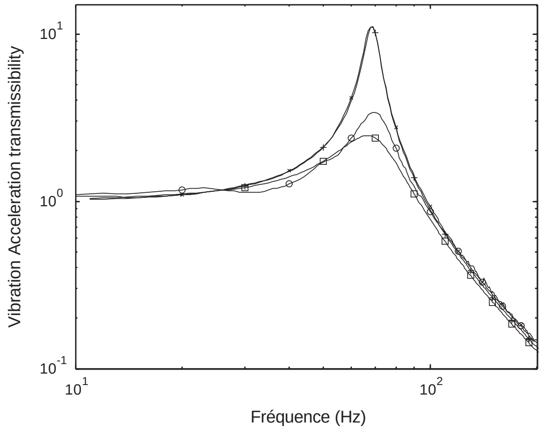

Fig. 12. Comparison of the mean measured and calculated vibration transmissibility response of a higher natural frequency suspended handle (four viscoelastic mounts, $K = 2.6 \times 10^5$ N/m, +, calculated without hand; ∗, measured without hand; □, calculated with hand; o, measured with hand).

6.3. Influence of the grip and push forces

Fig. 13 shows the influence of push force on suspended handle measured vibration transmissibility when the grip force is set at 50 N for the case involving the viscoelastic mounts suspension. Push force is observed to have little influence on handle transmissibility except in the 30–60 Hz frequency range where increased vibration attenuation appears to occur when increasing the push force. Within this frequency range, the influence of push force on vibration transmissibility appears to be most significant between 50 and 80 N, as compared to the range from 20 to 50 N.

Fig. 14 shows the influence of grip force alone on measured suspended handle vibration transmissibility when the push force is set at 50 N for the case involving the viscoelastic mount suspension. These results suggest negligible influence of grip force, provided that it is maintained within the 20–50 N range. For higher grip forces, some influence is noted on suspended handle transmissibility characteristics in the 30–60 Hz frequency range, possibly due to non-linearities arising in the viscoelastic mounts. On the basis of the results presented in Figs. 13 and 14, it may be estimated that, both push and grip forces are not likely to have any significant influence on the suspended handle transmissibility characteristics provided that the grip force is maintained within the 20–50 N range and the push force does not exceed 50 N. Similar trends were also found with the low natural frequency handle suspension which involved four helicoidal springs.

7. Discussion and conclusions

A model combining a suspended handle coupled with the hand-arm system was developed. This model applied the four-degree-of-freedom linear model defined in the ISO 10068 standard to represent the hand-arm system. This model was first validated alone on the basis of hand-arm

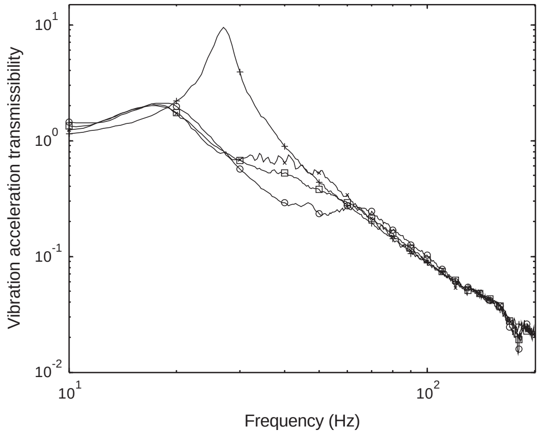

Fig. 13. Vibration transmissibility of a low natural frequency viscoelastic handle suspension, influence of the push force for a 20 N grip force (+, without hand; ∗, 20 N push force; □, 50 N push force; o, 80 N push force).

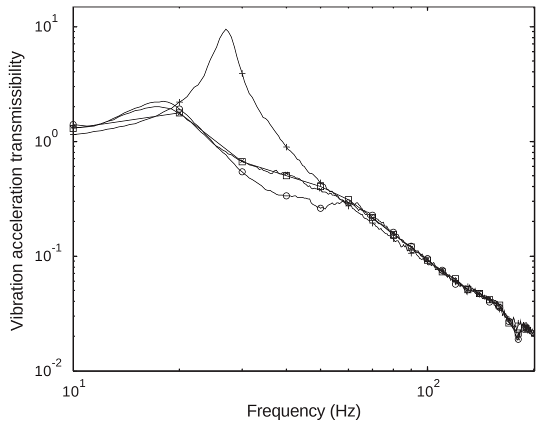

Fig. 14. Vibration transmissibility of a low natural frequency viscoelastic handle suspension, influence of the grip force for 50 N push force (+, without hand; ∗, 20 N grip force; □, 50 N grip force; o, 80 N grip force).

apparent mass biodynamic response measurements. Both measured and computed responses were found to be in agreement, particularly in the 50–400 Hz frequency range for the case where the push and grip forces were maintained close to 50 N. These results thus suggested that the hand-arm model could account reasonably well for the biodynamic response of the hand-arm system under the experimental conditions considered in this study and described in the ISO 10068.

A combined suspended handle–hand-arm system model was then developed and validated based on measurements of base to handle acceleration transmissibility frequency response functions determined for different types of suspensions.

For the free suspended handles having resonant frequencies tuned at 30 Hz, loading of the handles with the hand-arm system was found to introduce additional damping in the systems while shifting the resonant frequencies of the handles towards lower frequencies due to the mass increase provided by the hand at low frequency. On the basis of the model, it was estimated that the attenuation that could be achieved for both low-frequency suspended handles would be on the order of 30% at 35 Hz and 50% at 45 Hz, versus 0% and 50%, respectively for the free handle. In the case of a resonant frequency of the free system tuned at 67 Hz, loading of the handle with the hand-arm system was found to introduce additional damping alone at the resonant frequency, but did not change the resonant frequency of the system, due to the low value of the apparent mass of the hand-arm system at this frequency in comparison with the mass of the handle.

Two types of handle suspension were tested in this study: viscoelastic mounts and helicoidal springs. Although the vibration transmissibility characteristics were found to follow similar trends for both types of handles, the vibration attenuation provided by the viscoelastic mounts with and without loading of the hand was found to be slightly better. However, the use of helicoidal springs offers the advantage of behaving more linearly and being less sensitive to environmental parameters such as temperature or oil than viscoelastic mounts.

The influence of grip and push forces on the transmissibility response of the low natural frequency suspended handles was found to be relatively small provided that these were maintained within the 20–50 N range. Within the main excitation frequency range of a percussion drill (30–40 Hz), the influence of the hand-arm system was found to be relatively important on the vibration transmissibility characteristics of low-frequency suspended handles in view of the additional mass provided by the dynamics of the hand-arm system. In the design of portable power tools where the weight is a significant ergonomic parameter, the results obtained in this study illustrate the need to take the apparent mass of the hand-arm system into account to optimize the suspension characteristics needed to provide increased attenuation of the hand-arm vibration exposure.

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